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DRAWINGS ATTACHED.

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COMPLETE SPECIFICATION.

Improvements in Liquid Springs.

We, TAYCO DEVELOPMENTS, INC., a corporation organised under the laws of the State of New York, of the United States of America, of 188 Main Street, North Tonawanda, New York, United States of America, do hereby declare the invention, for which we pray that a patent may be granted to us, and the method by which it is to be performed, to be particularly described in and by the following statement:—

This invention is related generally to that class of device known as Liquid Springs, but, characterized by simple low-cost design providing reliable sealing means whereby said device lasts as long and can be produced commercially competitive with the plurality of coil springs they replace.

Many of the claimed or disclosed devices of the prior art have generally been drawn to the most critical of all elements related to such devices, that is, the sealing of a liquid in the spring. The remainder of the art relates to the design of structure to compress or contain high pressure liquid. For example, with regard to the former, it is known to employ a sealing method using a plastics tire seal for wear and sealing to provide the long leak-free life required to take care of the resiliency requirements due to deflection of relative components.

Another recent development involves utilizing plastics of teflon (Registered Trade Mark), nylon and elastomers, in combination, and called a chemical growth seal to provide the necessary seal compressibility pressure on the wall by the elastomer, nylon, teflon combination, from compatible chemical growth to take care of the deflection of cylinder walls while simultaneously providing a pressure in excess of maximum liquid pressure to seal the liquid whereby the seal always remains substantially in high pressure sealing engagement thereof.

There is also a known method of applying a plastics material to a spring member for the purpose of reducing the molecular gap between mating surfaces and providing the bearing characteristics between relatively sliding members whereby the seal can be supported and leakage can be eliminated. All of these concepts are indicated as a source of liquid spring improvements and, in fact, each concept referred to has improved the reliability and life of liquid springs but have heretofore not been elements of cost reduction so important for major fields, such as, the door closure, transportation, machine and die fields, etc.; where low-cost as well as long life are of paramount importance. This invention disclosed herein achieves both objectives.

Heretofore, there have been approximately five classes of seals used in liquid springs.

CLASS ONE: This seal has been the familiar "O" ring, Quad ring or Chevron elastomer which is intensified by internal pressure causing sealing elements to expand by internal pressure against the wall of the spring to seal and compensate for deflection and changes of pressures while the unit is stroked.

CLASS TWO: This sealing device has been that of the mechanical intensification seal first covered by Bridgman in American Academy of Arts and Sciences in which Bridgman accomplishes sealing because of the hydraulic intensification against a soft metallic flowable sealing element. This was largely a one operation laboratory sealing device.

CLASS THREE: This Bridgman seal

element was improved further by Dowty of England who employed a novel method using wire gauze or area reducing plugs of a metallic nature going through an elastomer and reducing its area while the part itself with the plug subjected the seal to intensification of a higher pressure whereby sealing occurs on a female seal on a shaft and the cylinder interior after the elastomer itself was initially preloaded. This seal has later been improved by the introduction of a teflon tire seal used internally on an external rod surface which has improved the wearing and friction characteristics of the Dowty seal. It has been suggested that a Dowty seal employing mechanically preloaded solid plastics, such as, nylon, teflon, and delrin (Registered Trade Mark) might also be used effectively, but all of these sealing elements comprise a relatively high-cost approach to the sealing of liquid springs for volume low-cost production.

CLASS FOUR: This relatively newer seal invention employs the chemical growth of an elastomer inside a liquid spring from an additive in the liquid and chemically accomplishes the intensification of seal pressure greater than the liquid it contains in a low-cost and relatively inexpensive manner, whereby the cost of the seals and springs and their reliability could be improved tremendously. However, the spring was still relatively high-cost as compared with the old familiar coil spring built on automatic machines. It is to the replacement of the coil spring of production by a low-cost liquid spring that this application is drawn.

CLASS FIVE: To which this patent application is directed embraces a seal design having a long leakage path or pressure gradient, plus employing a method of subjecting a single homegeneous intensifying seal to an elastic loading or interference coupled with a continuous intensification irrespective of temperature to which the unit is subjected, a seal which further replaces its own wear by mechanical cold flow of the plastics, to inhibit leakage through wear by mechanically replacing the seal material. It further provides a low-cost seal which, in turn, makes possible a low-cost reliable spring of few parts.

In the Bridgman or Dowty sealing configurations, area is removed from the seal to make it less than the intensifying piston area. In Bridgman, a differential area piston is used. In Dowty, pins are injected through holes in the elastomer which accomplish area changes. With seals of both types, however, costs are prohibitive except in high-cost applications. These seals are generally mechanically preloaded by mechanical compressors. A major problem with such a seal is that the variations in thermal expansion of the elastomer, as compared with the high strength steel cylinder, results in loss of preloaded pressure in cold weather and a high preload in warm weather, which, in the first instance, causes leakage and, in the second, seal wear. In addition, as normal seal wear occurs due to cycle life, the seal cannot replace its own wear except with service which restores the initial preload on the elastomer. The invention herein described is intended to also correct the above deficiencies.

It has long been felt that gains in cost reduction of liquid springs would come through the principle of cold extruding or cold working of steel of alloy nature for liquid spring cylinders. Using a huge hydraulic or compressibility press has indicated that if higher alloy steels in a ductile condition are struck with sufficient velocity with ounces of carbide or the other materials of sufficient strength to withstand a high impact load that the alloy steel can be made to work like tooth paste forming in one backward extrusion the shape of a liquid spring cylinder. Experiments with springs have shown because of the basic low piston mass situation related to the energy available from the compressed molecule that ram speeds of 200 inches per second are obtained. Such liquid spring presses have the capability to provide low-cost liquid springs by the extrusion of high strength allow steel members required for the cylinders. This is lower in manufacturing cost than present mechanical spring coilers. The present invention seal is intended to work with such extruded cylinders. Further disclosed are methods of cold working the cylinder to retain an end closure, preload stop, and volume reduction for accomplishing preload or leakage replenishment.

For short run special liquid spring or spring shock applications, a process of electrolytic honing has been developed. This method accomplishes the low-cost finishing of liquid springs by electrolysis with which this simple spring and seal configuration can be finished for seal compatibility.

In the invention disclosed here, a method is illustrated by which with this low-cost extrusion or honing, the sealing can be accomplished cheaply with very simple low-cost elements which combined with the method for extruding or finishing cylinders noted above provides methods whereby liquid springs can be mass produced as cheaply as coil springs and the degree of reliability with respect to sealing, makes them equal to many of the coil springs they seek to replace except in very longest of life applications to which coil springs are currently being subjected.

One of the principal methods by which this is accomplished is the replacement of the former complicated sealing element with

a simple plastics element which is caused to be the seal intensifier and wearing surface. This sealing element is adapted to low-cost molding or machining with high production equipment and when combined with a cylinder of proper stress levels for inward intensification, accomplishes the desired result of intensifying of pressure on a seal member whereby leakage cannot occur since its pressure at all times exceeds the internal pressure of the spring which it is sealing.

Since liquid springs and spring shocks have a universal market wherever springs or shocks are used, the concepts of this invention of low-cost sealing in combination with low-cost fabrication means are combined in endless combinations depending on the end market use. Thus, in die springs of 3/4 inch diameter, 2 1/2 inch length and .050 wall thickness, 2,000 pound springs, the low-cost seal combined with low-cost extruded cylinders, rolled retainers and ductile preloading are essential to that market which requires a low-cost throwaway simply serviced unit. However, in large aerospace requirements, say employing wall of 2 inch thickness and a 12 inch diameter of 1,000,000 pounds and combining the shock features the ability to service the unit to a higher preload say for a heavier second stage missile support requires the flexibility of variable loading on site and the ability to change dampening or liquids by dissembly and charging ports. Simple low-cost filling or charging means must herein be provided.

Also, certain zero impact springs, requiring low costs, utilize a slight modification of the seal piston combinations to utilize the seal elasticity as a counterbalancing spring to the liquid spring.

Similarly, when used as an automobile suspension, it may be desirable to maintain hydrostatic pressure on the low memory seal irrespective of the direction of shock absorber motion to reduce leakage in a low friction application.

A vehicle Liquid Spring system basically supports the vehicle on a 3/8 shaft 6″ long because of high liquid pressures of compressibility. Thus, the column strength of the small diameter shaft is critical. Bending or deflecting of the column must be followed by the seal or leakage will occur. This seal provides this feature.

It will thus be seen that the univeral need for a low-cost liquid spring in such widely divergent industries as dies, machine tools, transportation, health and medicine, aerospace, etc. will require a low-cost, reliable sealing means, though cooperating elements may of necessity be widely divergent.

According to the present invention there is provided a liquid spring including a cylinder, a cylinder end closure fixed to said cylinder and having a bore therethrough and an internal conical pressure angle, a seal member of plastics or co-operating seal members of plastics in interference seal fit with said cylinder or with said end closure member and having a bore co-axial with said end closure bore, said seal member bore having a lesser diameter than the bore of said cylinder end closure, said seal member having a conical complementing pressure angle in cooperation with the end closure pressure angle, and a piston member extending through said bores having a clearance or slip fit to said cylinder end closure bore and an interference fit to said plastics seal member bore.

The invention will now be more fully described by way of example and with reference to the accompanying drawings wherein:—

Figure 1 is a side elevation in section illustrating a combination liquid spring shock absorber embodying the basic features of the invention.

Figure 2 is a similar side elevation but showing the liquid spring shock under maximum shock load at end stroke illustrating the relative deflection and movement of the parts as well as the position therewith under maximum impact.

Figure 2a is a modified view of a variable flow dashpot head varying slightly from that of the device of Figures 1 and 2.

Figure 3 is a graph showing various shock curve features of the devices of Figure 1 through 4a and illustrating how this basic structure can be adapted to provide the variable conditions noted graphically in Figure 3.

Figure 4 is a side elevation in section of the basic structure as adapted to an extremely low-cost variable load liquid spring suitable for the tool and die trade and competitive, price and energy-wise, with coil springs.

Figure 4a is an end wall modification of Figure 4 after initial liquid loading, to accomplish a varying preloading externally.

Figure 5 is an illustration of the modified shock absorber of Figures 1 and 2 in its compressed position. The structure adapted to certain sophisticated vehicle suspensions where an extremely low hysteresis or seal friction is required and precise dampening is needed.

Figure 6 shows the various elements of Figure 5 in an unloaded or preloaded position.

Figure 6a illustrates an enlarged fragmentary detail of a low-cost epoxy cementing of relative part of this liquid spring.

Figure 7 is a side elevation view of a liquid spring adapted to extremely long storage life on the shelf with relatively low cost parts providing a greater intensifying seal element.

Figure 8 illustrates a further modified unit adapted to provide a tension-compression configuration for zero impact conditions on machine tools or impact shock buffers.

Figure 9 illustrates graphically the low impact balanced spring condition of the device of Figure 8.

Figure 10 illustrates in detail how the devices with a replenishing feature can be filled externally through a single opening in a liquid spring chamber, and the apparatus required for this.

Figure 11 is a sectional view taken along line 11, 11 of Figure 10 illustrating how relative units can be rotated or retained against rotation.

Figure 12 is a sectional view illustrating how the life of the liquid spring is restricted when the surface finish of the relative moving elements is not in the direction of travel.

Figure 13 illustrates how, when the direction of surface finish is correct, abrasion and tearing off of the structural sealing element does not take place.

Figure 14 is a further modification of this liquid spring to provide high preload pressures when an extremely high spring rate with a short stroke is desired.

Figure 14a is a modification of the device of Figure 4 to provide the same high preload situation in a low cost spring structure.

Figure 15 illustrates yet another means of preload retention of the basic structure disclosed heretofore, in which the piston preload holder is a tension element threaded into a section which is also utilized for retention of the orifice ring of Figure 1 and 2.

In Figure 1 and 2 is illustrated a liquid spring shock 20 of novel construction employing a new long pressurized leakage path seal 50. Liquid spring assembly 20 employs a cylinder 21 having a cylinder wall 23, an endwall 22, and an externally threaded extension 24 having an internal cavity 24a which has a thread 24b at one end or full length for the retention of the metering ring 25 or preload stop 890, (Figure 15). Ring 225 is intended to cooperate with metering probe 41 of the dashpot head 40. Dashpot head 40 is further characterized by a second orifice passage 42 of annular construction and an annular liquid passage groove 47 with a series of peripheral holes 43 for taking metered liquid from annular groove 47 into the seal cavity 53 there behind. Metering is accomplished in this instance by the passage of liquid through metering annulus 42 for, as the piston is stroked to the compressed position of Figure 2, shock response is influenced by the tapering cylinder wall 23a which provides desired orifice restriction with reduced velocity. As seen in Figure 2a, a yieldable split metallic or solid plastics piston ring 44 on dashpot head 40 provides a means for sealing the member on the tapered wall while simultaneously preventing binding of the piston as it goes down tapered wall 23a.

The tapered wall 23a can be precisely developed by the method of electrolytically honing cylinder walls by a graded exposure or tapered anode. Still another method is that of forcing the cylinder into an internally tapered holding cylinder then boring and honing mechanically to provide a straight wall while the cylinder is in forced deflection. When the straight walled cylinder is removed from its fixture, it then returns to its former configuration, thus providing an internal tapered wall. This then provides a metering configuration in a liquid spring as it is stroked providing shock characteristics, such as curve C Figure 3 or as it is desired according to the metering configuration. By choice of materials having different coefficients of friction and elastic deformation, friction dampening can be superimposed on hydraulic dampening. For instance, dampening curve D can be superimposed on shock force "C" as a secondary friction force. A third shock force can be accomplished by the probe 41 entering the metering ring 25, at end stroke, as in Figure 2, and as shown in Figure 3 graphically as Curve E. Curve B of Figure 3 is the liquid spring curve due to compression of the liquid. Thus, the curve A represents the actual spring shock curve, the sum of Curves B, C, D and E. Dashpot head 40 has a coaxial shank 46 threadedly engaging bore 35 of a liquid spring piston member 30. Piston member 30 is characterized also by a threaded section 34 which receives a threaded section 49 of the dashpot head to hold the assembly 30, 40 together. Piston 30 has a refilling or servicing bore 31 which terminates into a smaller bore 33 and a valve or seal 32 between the threaded element 49 and a sealing face 36 at the bore 33 to permit sealing of the bore as piston member 30 is threaded down after pressurizing, to seal it much in the nature of an automobile valve cap as will be described hereinafter in Figure 10 with respect to filling methods. This design limits openings in the high pressure chambers to one opening carried remote from the pressure wall for maximum availability and uses dashpot head 40 as the check valve at lip 56, as filling is accomplished.

Sealing in this particular spring of Figures 1 and 2 and 4 is accomplished with a plastics yieldable member 50 of teflon, nylon, or delrin having an annular groove 53 formed therein providing two lips 56 and 57 for sealing the piston 30 and the cylinder surface 23a respectively and providing a long leakage path bore 58. Plastics 50 can be of delrin, teflon, or nylon or any

good bearing and flowable plastics which incorporates a long leakage passage 58 and makes an interference fit on the internal diameter of the cylinder wall 23a preferably by at least 0.15 per inch of diameter. This then provides a squeeze which is resisted by the cylinder wall 23, which is caused to deflect .001 per inch of diameter to balance the pressure by yield outward and provide with its hoop tension an elastic pressurized condition on the external surface of plug 50 at all times. This small deflection provides a large volume of replenishing from cold flow at piston 30 as it wears. Lip 57 acts to provide an elastic internal preloading on the piston 30 to eliminate initial leakage until the unit is pressurized by its own internal pressure to cold flow the plastic to fill minor surface finish irregularities on the piston surface 37. Lip 57 and 56 act as flexures to follow any slight side load deflection or misalignment of piston 30 to assure continuous pressurization. A solid piece of teflon would tend to gap or elongate on side load and seal leakage would begin immediately.

To assure leak free seals over long life, cold extruded cylinders finished linearly in the direction of travel by the surface finishing method hereinbefore referred to are used. With cross hatch or ground surfaces not parallel to seal travel, the plastics would wear rapidly at high pressures as will be discussed in Figures 12 and 13.

In Figure 2 is shown the compressed position of the spring shock of Figure 1 illustrating in dot dash lines how the cylinder is deflected in this case being in the neighborhood of approximately three thousandths at the middle portion and about a thousandth and a half at either end on a unit 1 1/4″ diameter 3 1/2″ long. It should be noted that this yielding provides a larger piston area at the lip 57, than is accomplished at its conical base 68 always insuring seal intensification. As noted in Figure 2, the energy storage and movement of the seal 50 is shown dotted at 56A, 57A, 53A wherein the seal has compressed 6% at 20,000 P.S.I. due to its compressibility. This working of lip 57A requires that the cylinder wall 23 be linear lapped, cold-extruded or otherwise furnished with linear finish lines to prevent seal abrasion and leakage. Figure 2 further indicates that both metering orifices 42 and that at 25 are now tightly closed off providing a shock curve A generally flat topped but with a peak at the end as shown in Figure 3. This is similar to the automobile bump configuration provided by the rubber bumpers at the end of the automobile shock absorber stroke.

The long leakage path has been tested as a means of sealing, and develops a pressure drop like a labyrinth seal by virtue of its length. A common teflon slug has been tested which will seal until it cold flows into a less elastic mass about 50,000 reversals. Conversely, lips 56—57 remain in elastic spring loaded interference with piston 30 cylinder bore 23A for 750,000 cycles of leak-free operation. The pressurizing feature of the liquid plus the interference of lips 56—57 and squeeze at 53 due to the elasticity of the steel cylinder and the 30 degree pressurizing angle 68 of the end closure 69 all cooperating for long seal life. As plastics seal member 50 is driven up angle 68, it replenishes its wear for perfect leak-free long-cycle life. It should be noted that plastics 50 can be caused to flow into guide cavity 65 of end closure 69 as shown in Figure 13 to provide bearing for side loads on piston 30. This same flowable condition down the angle 68 also causes seal replenishment due to wear with the U-type integral pressurizing piston 50 having lips 56—57 and cavity 53. The seal and integral pressurizing piston 56—57 and cavity 53 act to cold flow the plastics at bore 58 to always assure a tight sealing surface in greater interference pressure-wise than the liquid it contains. The pressure angle 68 being 30 degrees the seal 50 exerts a sealing pressure at bore 58 up to a maximum. 1.6 times the liquid pressure due to this pressure angle 68. This seal then maintains an initial pressure because of the interference fit in cylinder wall 23 and piston 30 and sealed in a lip 56—57, and an intensifying seal pressure because of the angle 68, thus providing 750,000 leak-free cycles for liquid springs. This life is 1/3 greater than the life of a rear leaf spring of a vehicle under maximum cycles and 12 times that of a vehicle shock absorber.

Figure 13 is a greatly enlarged fragmentary view of this seal area showing the plastics 50 extruded into the guide clearance 65 with linear finish 30 on the piston 30. In Figure 12, the extrusion of the plastics 50 into the guide clearance 65 is breaking up and disintegrating seal 50, forming a poor seal because of the circumferential finishing 30″ of the piston 30. These are actual test results where linear finishing lasted 7 to 1 over circumferential finish.

Cap 69 can be of a structural bearing alloy, such as, beryllium copper or Navy Phosphor bronze for high cost aerospace uses of liquid springs to avoid scoring the piston rod 30 in the event of a side load, piston buckling tendencies or deflection. It can have a silver or copper plated bore.

For low-cost springs, a permissible plastics cold flow such as at Figure 13 is desirable. It has been noted that teflon will cold flow to a specified position orienting its molecules to increase its own strength where-

by its resists further cold flow. This produces a low cost, high strength barrier to further cold extrusion of the softer teflon from which it cold flowed its high strength section. To avoid scoring the piston on excessive side loads, an exit venturi like shape 65a can be employed. This permits seal 50, piston 30 to deflect, or tend to buckle slightly which the seal can accommodate, so the hard piston 30 will not score by rubbing contact on cap 69 and the lip 50a acts as a wiper for the piston 30.

While the seal 50 is shown in contact with the cylinder 23a, it should be understood that seal 50 can be recessed in a solid end wall of a much larger cylinder where it would only be in contact with a seal bore or recess. While such a structure is not shown, it is obvious that so long as the proportions of seal length, angle and lips are followed the structure shown will function equally as well in such a large spring. It should be noted that 90% of the production is in springs of less than 3" diameter so that in most instances the seal configuration shown in the drawings, where the seal rests on the cylinder wall, would most generally be used.

The seal principles disclosed herein have the features of a three element seal in a single homogeneous molding or machining of plastics. One section being a flexible "U" type cup seal but with long thin flexible lips elastically yielded in sealed contact to the piston and cylinder, integral with a body section in actual interference fit with a seal cavity or cylinder and the piston, terminating in a third conical section adapted to wedge sealing and seal replenishment from internal pressure.

In Figure 2A is illustrated a version of dashpot head 40a characterized by a plurality of orifice holes 47a which is controlled as to flow by the ring 44 which as it is reduced in diameter restricts the orifice flow through holes 47a by reducing their cross sectional area as ring 44a is reduced in diametrical size by traveling down taper 23a.

Figure 4 is a version identical to that of Figure 1 and 2 except that it is just a liquid spring providing the curve B of Figure 3. The primary difference is that this piston 430 is a single hollow rivet having a riveting extremity 436 with a shank 434 and base 435 which is riveted over at 437, a stamped preload retention collar 440 having a drawn bore and shoulder 446 engaging shank 434 and formed seal engaging stop 447. This then produces an extremely low-cost piston assembly for a low-cost, pure liquid spring, such as, for the tool and die trade. It should be noted that stamping 440, piston 430 provide an unused space gap 470, on piston 430, which provides a section of piston 430 for holding when superlinear-finishing and for possible distortion on riveting. An anti-extrusion collar 451 of high strength plastics or bearing alloy is employed to assist in preventing any loss of seal 450 over long life. Since it is a more spring-like material, it can have greater interference on shaft 430 and act as a wiper for external dirt on shaft 430. It also prevents abraded seal particles from escaping to reduce spring forces.

Cylinder 420 is caused to be yielded by rolling a retaining collar or groove 425 into groove 463 of the cylinder end closure 460 to retain it herewith. This construction is about 1/3 the cost of precision threading the respective elements. This also acts to retain a spanner wrench, engaging holes 426 for threading 420 into a socket.

Since positioning of end closure 460 is difficult with respect to liquid volumes and thus preloads on piston 430, the present embodiment proposes to yield the stud end 480 at 490 by impressing with a ball or ball end tool to get a specific preload on piston 430. This can be used to develop specific preloads initially or after cycle life by the choice of balls and the depth. Thus, any toolmaker can increase the preload without risking seal leakage. The wall yields mechanically cold to hold 20,000 P.S.I. without losing its mechanical set.

Figure 4a illustrates that rolling the cylinder 420 at 491 can also accomplish this. In both instances, metal having a yield of 150,000 P.S.I. is yielded providing a constant high pressure source to the seal 150 to keep it intensified in sealing contact at 190 with piston rod 130.

Figure 5 illustrates another version of this device at its end-stroke indicating how the metering is closed off. It should be noted that this seal 150 with its long seal lips 156, 157 provides low friction characteristics with a pressurized long leakage path for very low friction and long life. Since in this unit interference at piston surface 190 is minimized to provide low friction, such as for a long stroke vehicle suspension, we have utilized a nylon anti-extrusion member 158 which is not subject to the angular intensification of angular anti-extrusion member 451 of Figure 4 but still prevents extrusion. Although this does not appear from the drawings the bore through enclosure 160 is, of course, of larger diameter than the bore through the seal 150 to provide a sliding fit.

It is well to note here that in an actual automobile shock absorber having a 6" stroke and providing a supporting spring load of 800 pounds and a shock force of 2,000 pounds per wheel with a spring rate of 90 pounds per inch, the column strength of piston 130 is critical. High strength elastic metal alloys will just accomplish this

on a 3/8 shaft 6" long without permanent buckling. However, column deflection and rippling and bucking tendencies are to be expected from shock loads. In an existing shock absorber, for such use, the shock shaft is generally one-half inch in diameter and a very soft elastomeric seal is used to follow such shaft deflections. The more rigid plastics used here would customarily not follow except for the long lip 156 which is in elastic interference with the shaft at 151 and follows it through such deflections. Actually, the solid section at the radiused and conical angled surface acts as a ball joint to allow this slight bending or deflection of piston 130 while still sealing it tightly.

Actually, this is an important feature of seals 50 and 450 in Figures 1 through 4, 14a and 15 in that even short stroke slim pistons have this deflection and harmonic rippling or vibration on impact which must be accommodated by the seal. Thus, the solid and conical seal section behaves like a ball joint for minor deflections while the lips 156—157 follow the deflecting rod.

In Figure 5, cylindrical shells 125 and 141 are attached by one end each to the seal 150 and the piston 140 respectively, and telescope the former inside the latter to form an inner chamber communicating through damping ports 143 with the piston face.

Figure 6a is an enlarged fragmentary view of the seal end of the liquid spring 120 showing the epoxy resin 121^1 confined between a counter bored are 121^{11} of the cylinder and an enlargement 160^1 of the end closure 160.

In Figure 7 is shown a pure liquid spring 220 without the shock characteristics of metering and characterized by the piston preload device 240 which is, in effect, similar to dashpot head 140 and 40 except that it does not have any metering portion. Preload device 240 has an upstanding shoulder 241 and annular flange 241^1 bearing on sealing configuration 250. This sealing configuration or composite member 250 comprising an outer sealing wedge member 251 and an inner sealing member 252 having tapered surfaces 253 and 254 respectively, said sealing surfaces wedging against the steel outer cylinder 221 and against the piston 230 along leakage path 290. This device has the characteristic of intensifying pressurization of the seal at all times against the piston shaft 230 and the wall 221 by wedging action as deflection and wear takes place along leakage path 290 and assures at all times an interference on piston member 230 in excess of the liquid pressure in the chamber. Further modification of this wedge type part providing the sealing is characterized by the reversal of the angle. If the angle of wedge is 30° as in Figures 1, 2, 4 and 8 the ratio of piston seal pressure to liquid pressure is 1.6 to 1. If 45°, as in Figures 5 and 6, it is 1 to 1. If 15° as shown in Figure 7, it is 2.1 to 1.

In Figure 8, the seal having the pressurizing angle bears against the end member 351 which can be the steel end closure or a separate plastics piece. Initial and intensifying pressurization is through means of an "O" ring elastomeric seal 352 formed in a retention groove therein. This device is further characterized by the ability to provide wear replacement as it extrudes to replace wear from the bore 353 against piston 330 at all times. It should be noted that face 341 of head 340 is impressed into elastic plastics seal member 350 to provide an elastic balancing spring condition whereby slight tension loading can be accomplished or zero impact condition in event of an impact condition in compression. Such a shock condition is desirable in a preloaded liquid spring. Slope F of the graph of Figure 9 illustrates this principle. Slope F^1 illustrates how a tension load is handled elastically by seal compression.

In Figure 10 is illustrated means for filling the devices of the previous drawings in which a filler cap 300 having a bore 301 and a teflon guide seal 302 is caused to fill through the piston members 30, 130, 230, by sealing contact thereof and to retain presure as the pressurizing liquid flows through the valve element of pistons 30, 130 and 230, elements are shown with the pistons 30, 130, 230 threadedly backed off from the end stop member 240 that acts as a check valve to prevent the liquid normally from going out of the cylinder. Pressure therein enters through the bore 431 of piston members 30, 130, and 230 and unseats the safety teflon cap seal 432 forcing the liquid down past the threaded elements and lifting the pressurizing cap off its seat and permitting the preloading of the unit to any desired pressure. It will be noted that this can be done with a separate element or it can be done with a unit which is thereafter caused to engage the spring as a separate entity. Referring also to Figure 11, a downwardly formed lug 543 on cap 540 engages an interruption 550^1 in the annular groove 555 in seal 550 to hold it against turning like a wrench does while piston 230 is turned by a strap wrench.

As discussed hereinbefore, Figures 12 and 13 illustrate a very essential ingredient of the piston 30, seal 50 retainer 69 combination. Liquid spring pressure against seal 50 causes it to wedge against surface 68 assuring a higher pressure between seal 50 and piston rod 30. This, therefore, then eliminates leakage. However, semi-rigid

plastics, such as, teflon have a tendency to cold extrude and abrade because of their softness. In the spring shown in Figures 1 and 2 and others a circumferential finish shown by the lines 30[11] chews the seal up and extrudes it through cap 65 as shown in Figure 12. This is true even though the surface finish is under .05 micro finish. The spring under such configuration is rendered useless under 100,000 cycles. Referring now to Figure 13, the teflon gland 50 has cold extruded partially into groove 65 of cap 69 and formed a thin high strength lip 50a which with the piston rod linear finished as in 30[1] does not abrade. A sealed piston rod conception of this configuration has reached 750,000 cycles or 7 to 1 over that of the 100,000 cycles of Figure 12.

In Figure 14 is illustrated a method of obtaining a high preload by forming an upstanding shoulder 667 on cap member 668 which retains seals 650. Supplementary elastomer 669 seals cap 668. Shoulder 667 bears on piston head 640 to take high preload which the plastics 650 will not contain without extrusion.

Figure 14a illustrates in a fragmentary section, a preload retention device 767 which has a circumferential groove similar to the manner cap 450 is retained in Figure 4. A similar yielded cylinder head 725 holds cap 767 against the preload head of piston 740. Still another method of holding high preloads is illustrated in Figure 15 in which a piston head 840 is herein retained by tubular stop member 890 which threadedly engages cylinder 720 at thread 724 with its threads 891 being adjustable therein to provide for specific length on piston 720. A plurality of orifices 893 permit exhausting liquid from chamber 895 and allow flow therethrough.

WHAT WE CLAIM IS:—

1. A liquid spring including a cylinder, a cylinder end closure fixed to said cylinder and having a bore therethrough and an internal conical pressure angle, a seal member of plastics or co-operating seal members of plastics in interference seal fit with said cylinder or with said end closure member and having a bore co-axial with said end closure bore, said seal member bore having a lesser diameter than the bore of said cylinder end closure, said seal member having a conical complementing pressure angle in cooperation with the end closure pressure angle, and a piston member extending through said bores having a clearance or slip fit to said cylinder end closure bore and an interference fit to said plastics seal member bore.

2. A liquid spring as claimed in claim 1 wherein an annular anti-extrusion ring is provided in juxtaposition with said cylinder end closure.

3. A liquid spring as claimed in either claim 1 or claim 2 wherein the sealing member bore engages said piston member for a length at least equal to three times the diameter of said piston member.

4. A liquid spring as claimed in any of claims 1 to 3 wherein the plastics sealing member has an inner sealing lip engaging said piston member and an outer sealing lip engaging said cylinder.

5. A liquid spring as claimed in claim 4 wherein each of the sealing lips has a substantially constant thickness along its length.

6. A liquid spring as claimed in claim 4 or claim 5 wherein each of said sealing lips is relatively thick along its entire length.

7. A liquid spring as claimed in any of the preceding claims wherein said plastics sealing member includes a section between said conical face and said sealing lips which is in interference fit between the piston rod and the cylinder.

8. A liquid spring as claimed in any of claims 2 to 7 wherein the anti-extrusion ring has a conical face engaging said end closure conical face where the rod passes through a bore in said end closure.

9. A liquid spring as claimed in claim 8 wherein said anti-extrusion ring is fabricated of a plastics material which is structurally denser and stronger than the plastics material of said sealing member.

10. A liquid spring as claimed in claim 8 wherein said extrusion ring is fabricated of bearing metal.

11. A liquid spring substantially as herein described with reference to the accompanying drawings.

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London, W.C.1.

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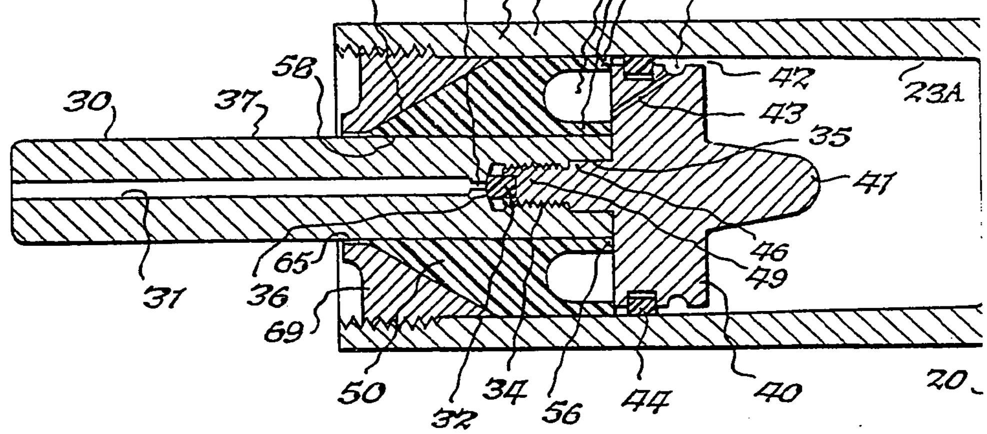
Fig. 1.



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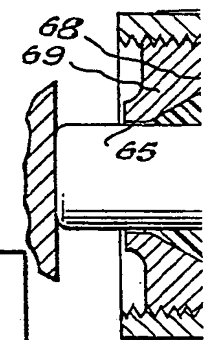

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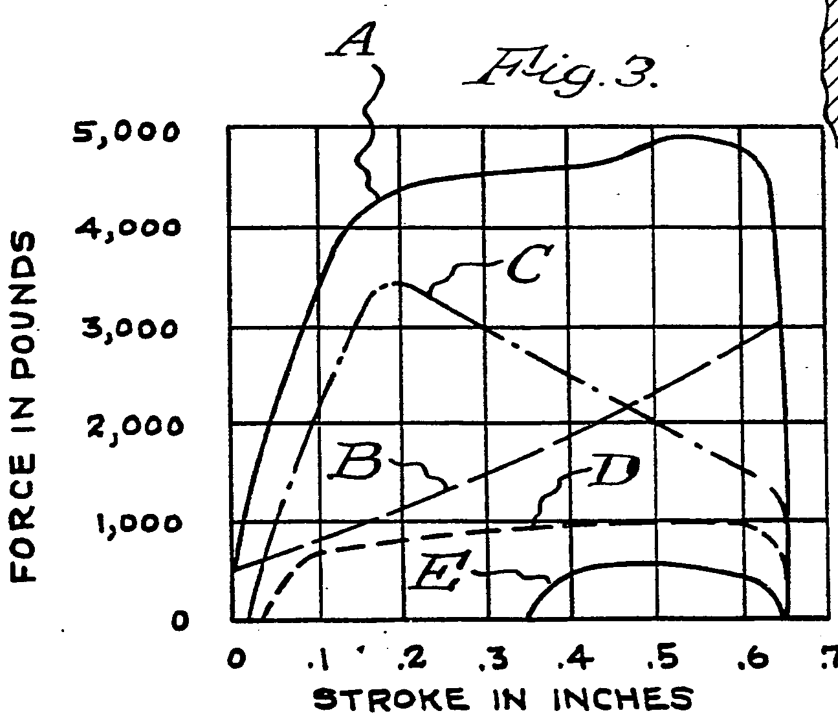

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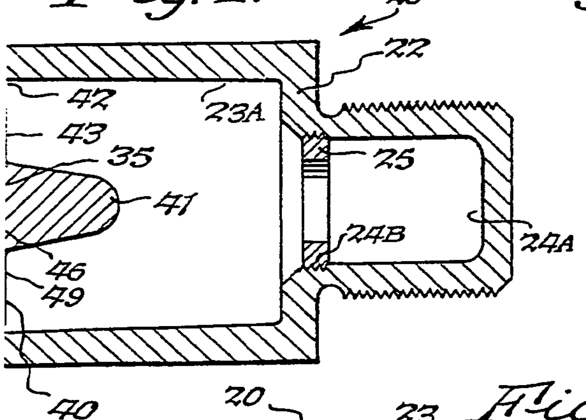
STROKE IN INCHES

Fig. 1.



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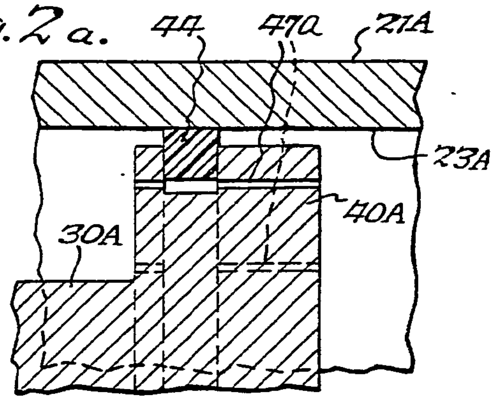

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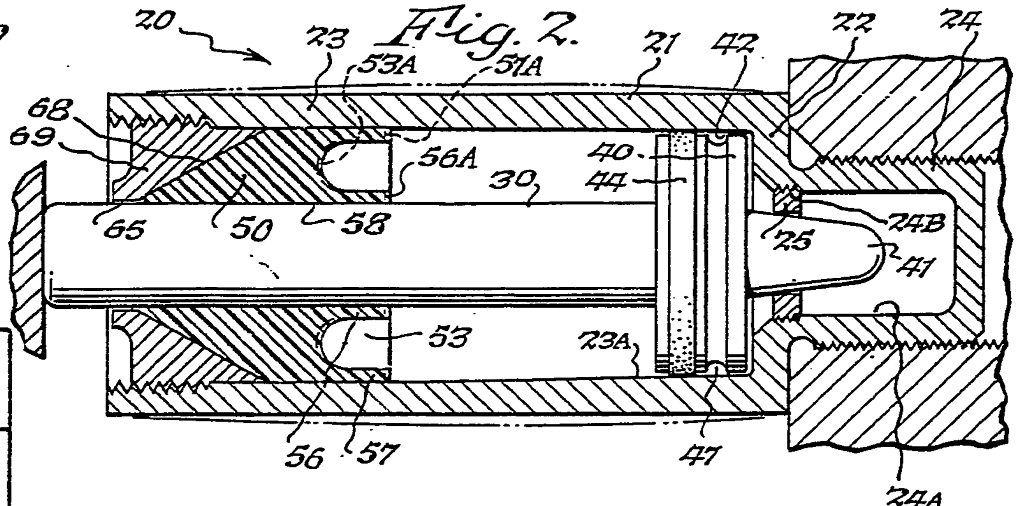

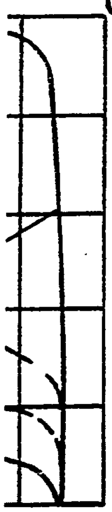

.6 .7

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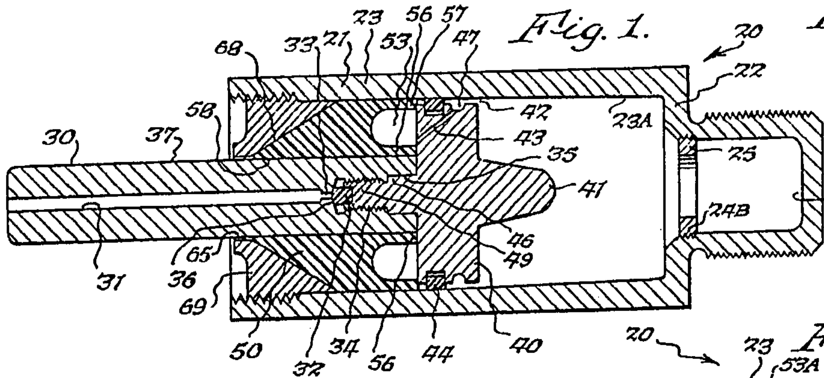

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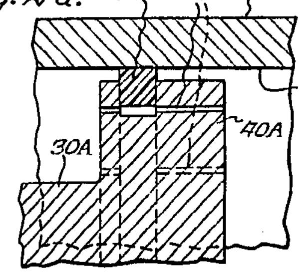

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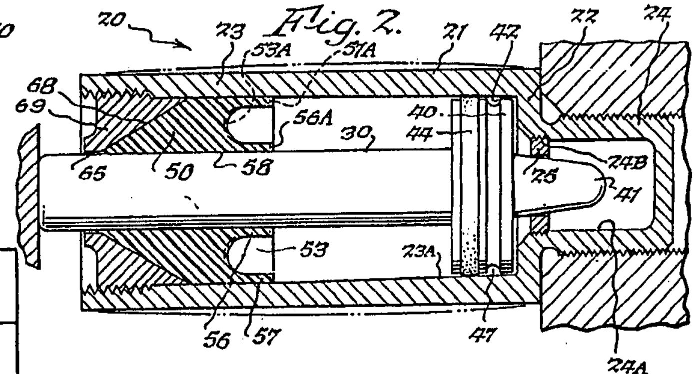

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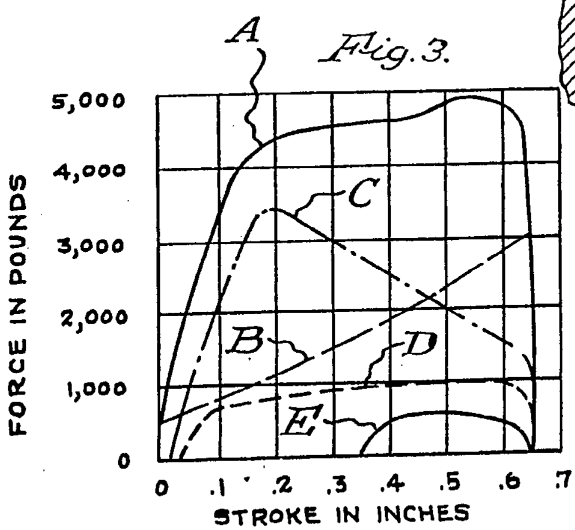

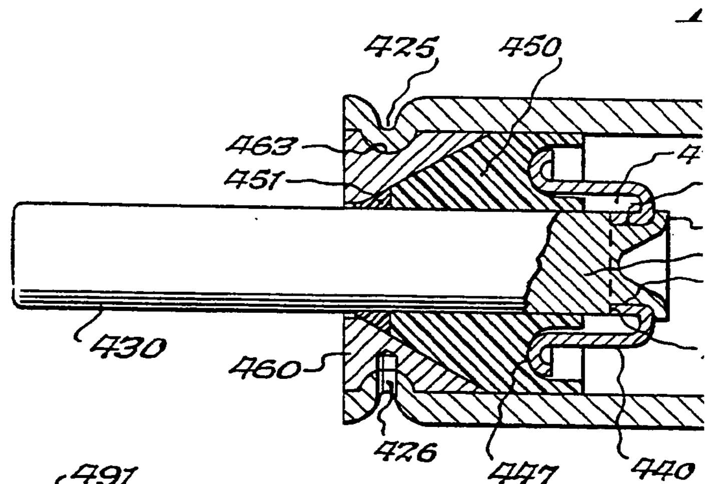
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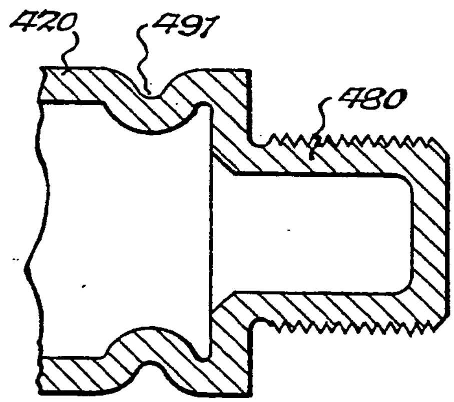
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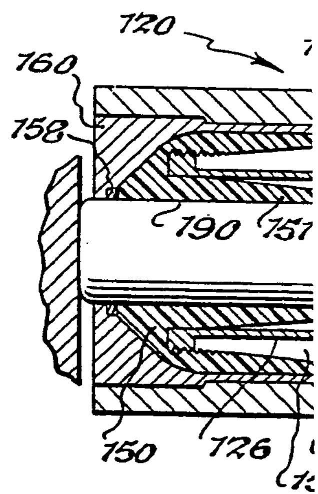
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1069895 COMPLETE SPECIFICATION

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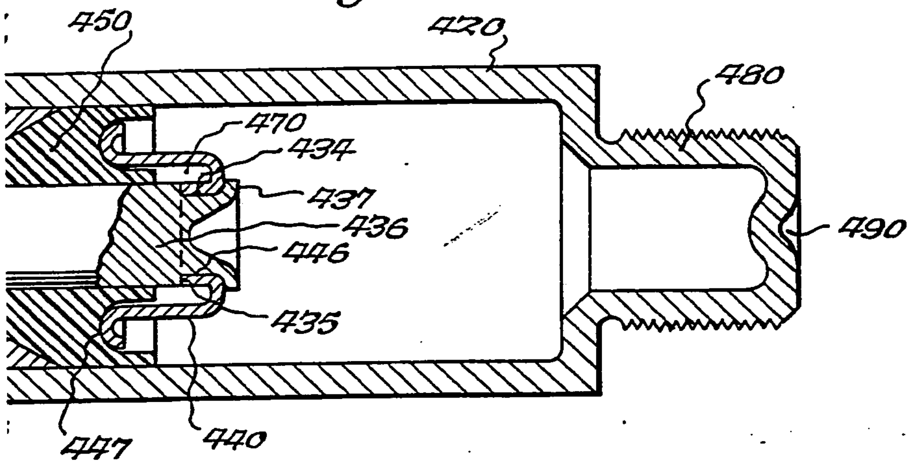
Fig. 4.



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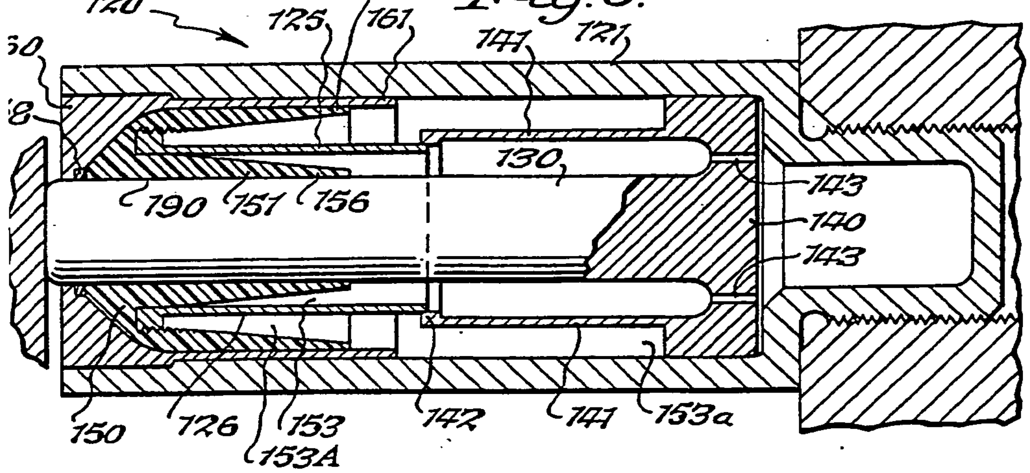

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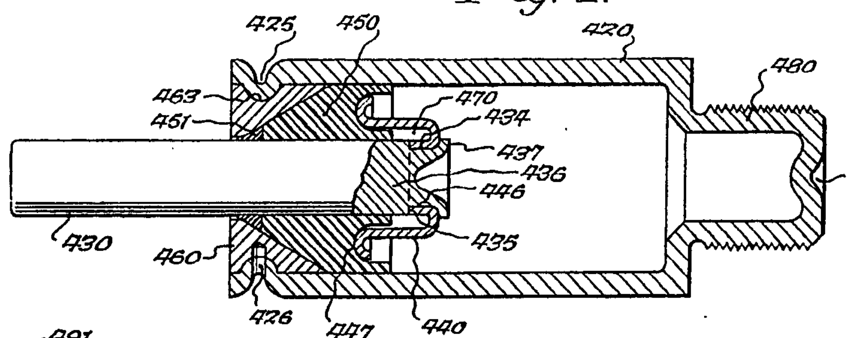


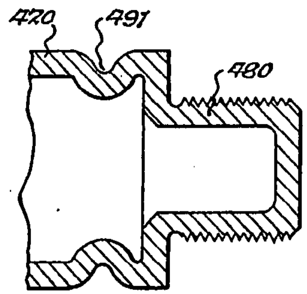
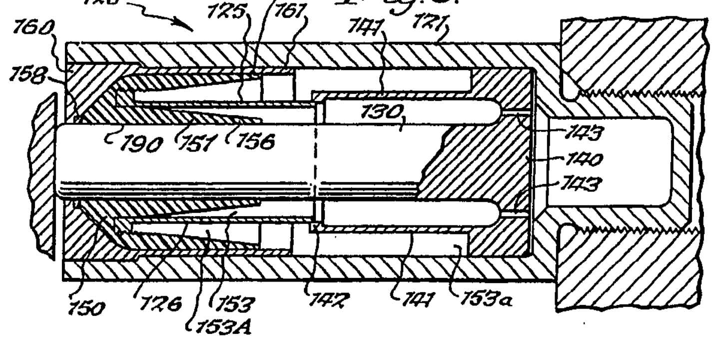


Fig. 4a.

Fig. 5.



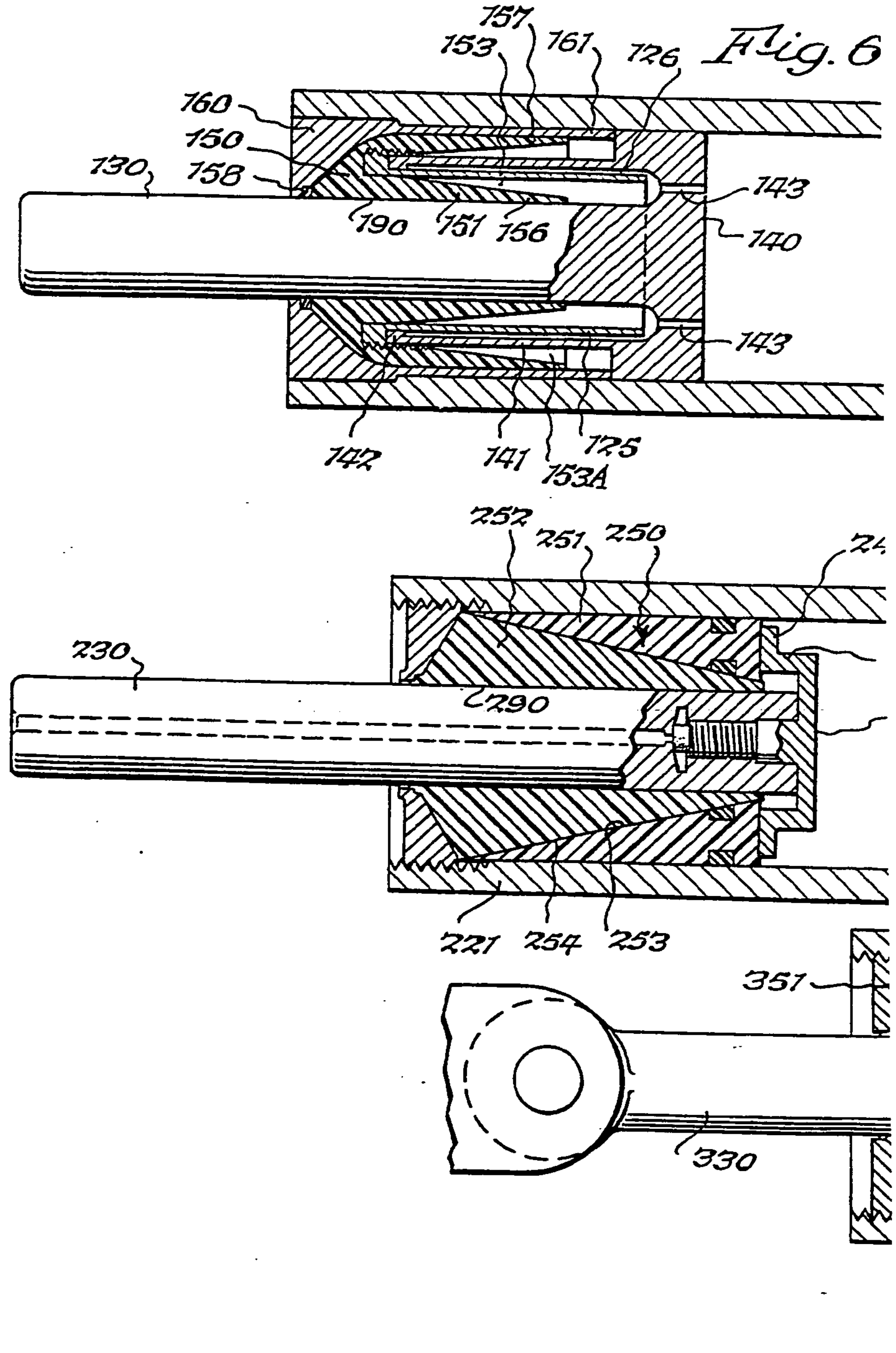


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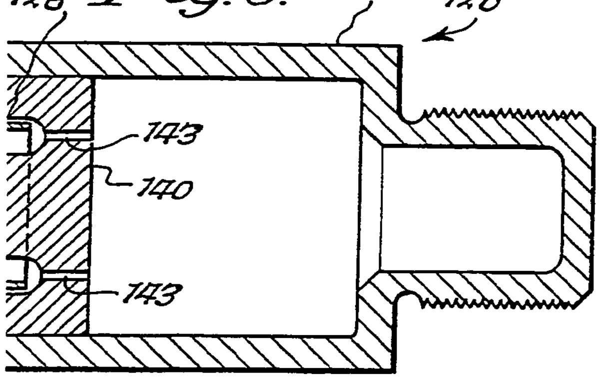
Fig. 6

Fig. 6.



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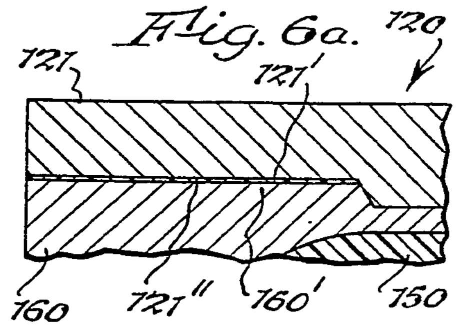

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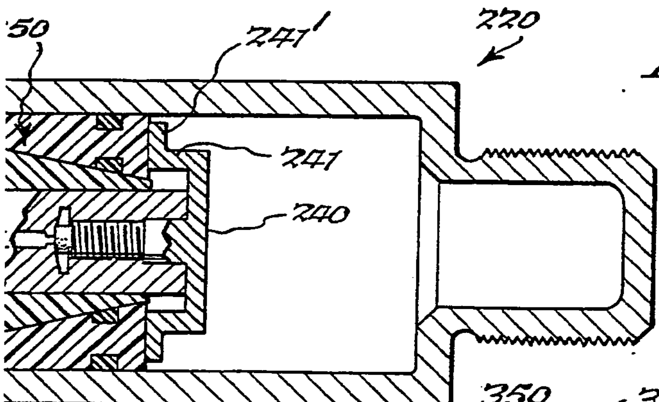

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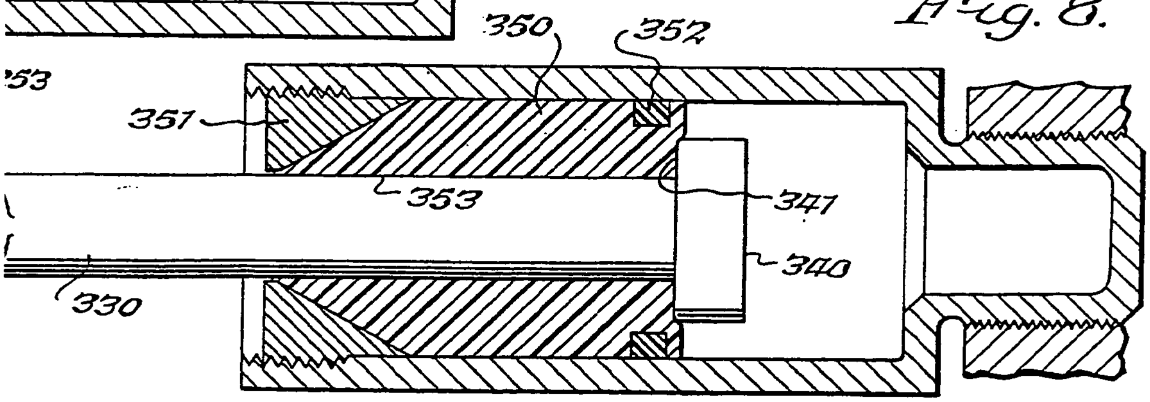

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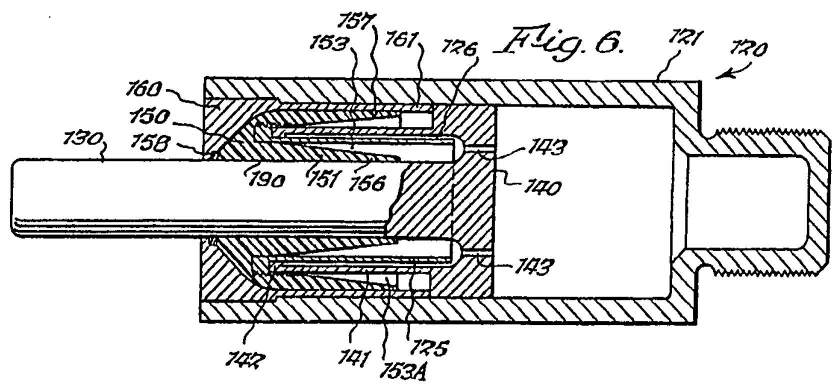

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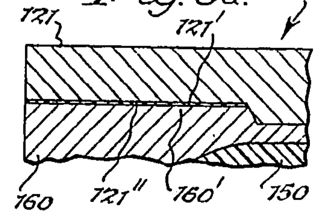

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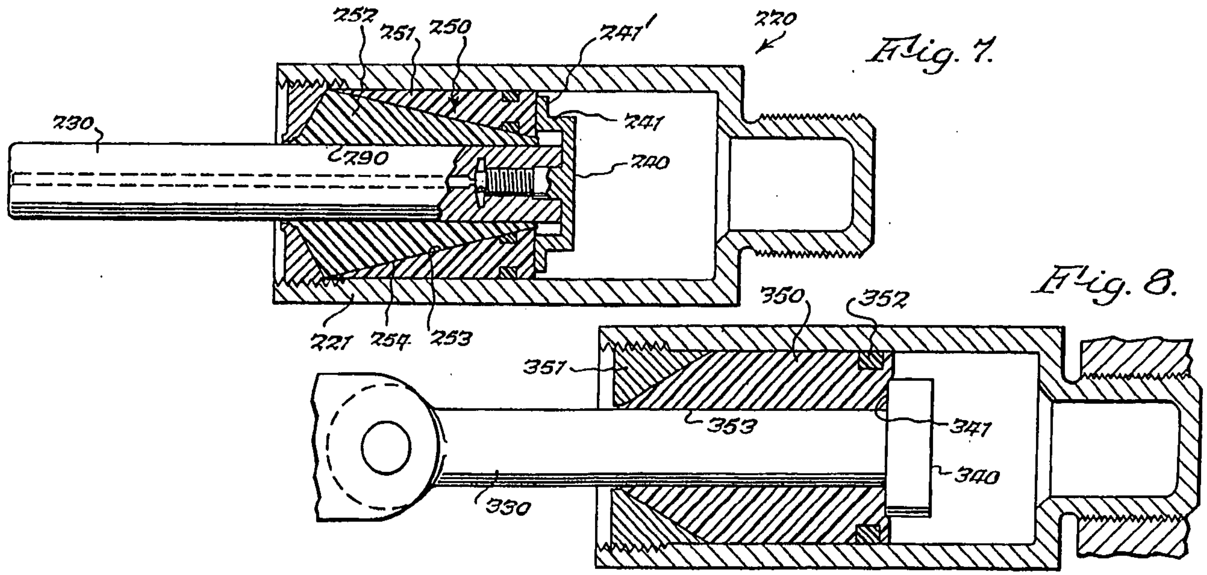
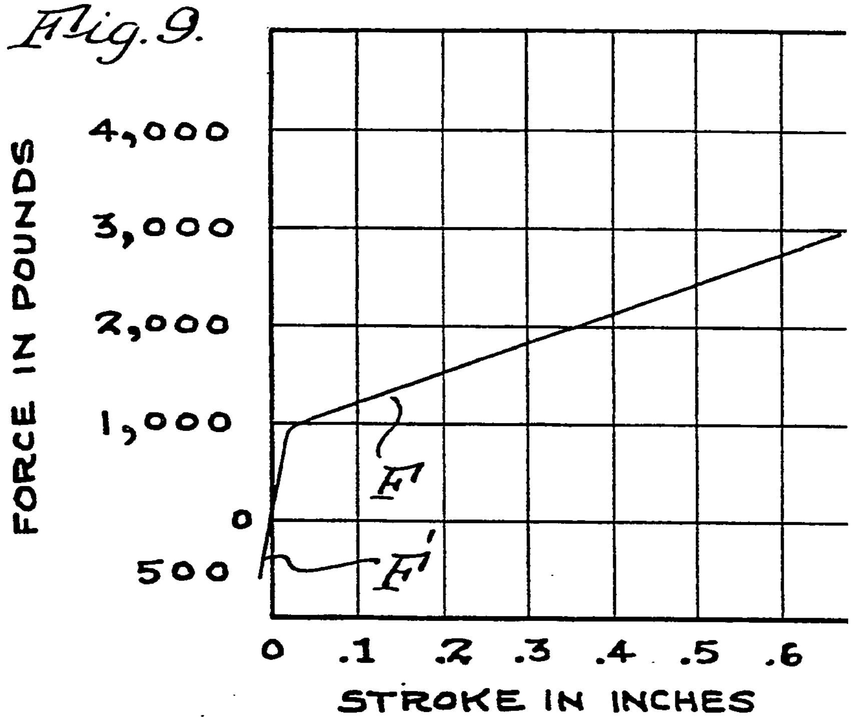


Fig. 7.

Fig. 8.

Fig. 9.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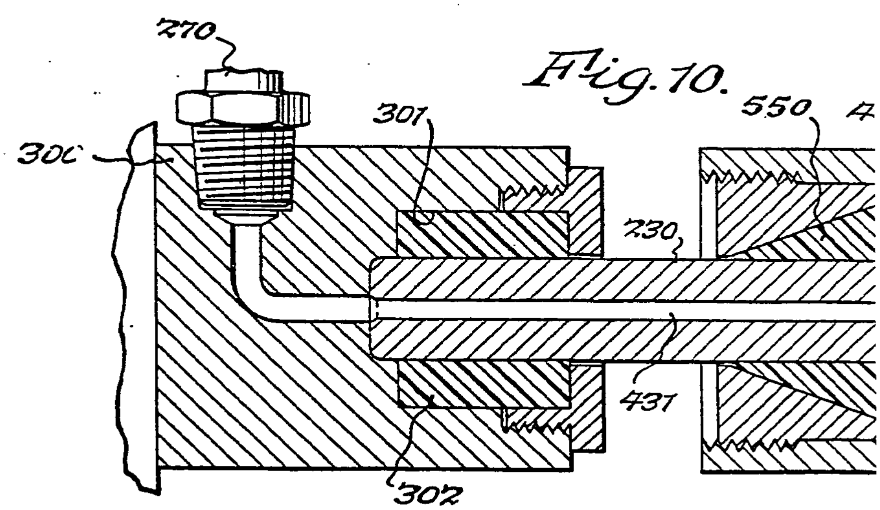
STROKE IN INCHES

Fig. 10.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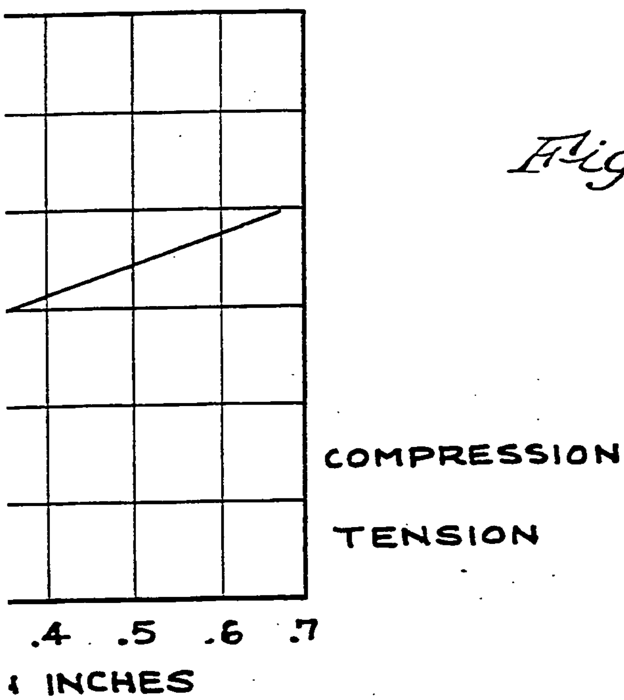

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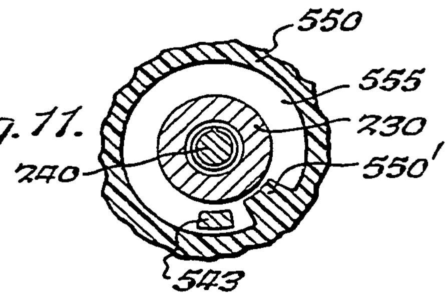

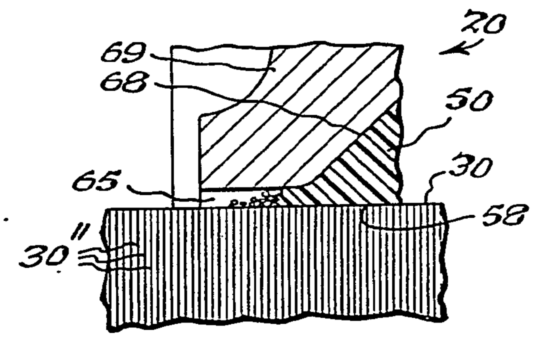

Fig. 12.

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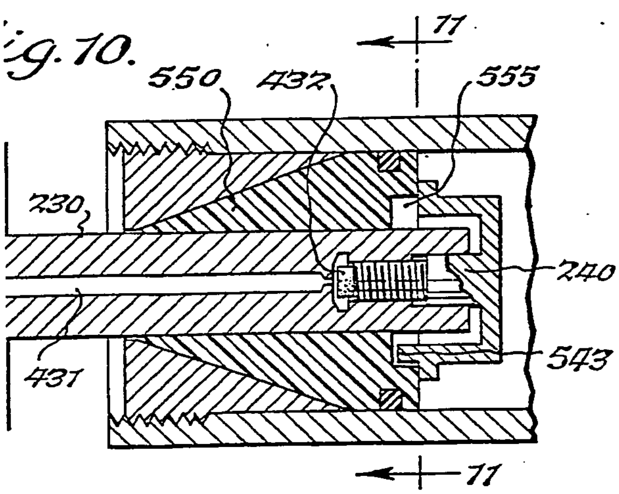


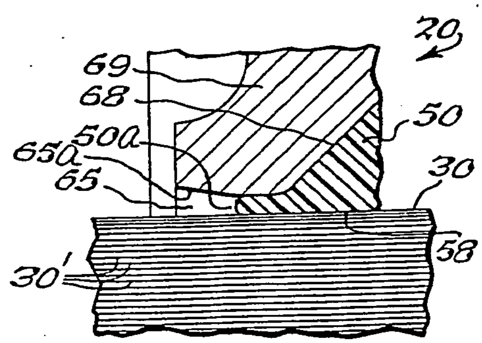
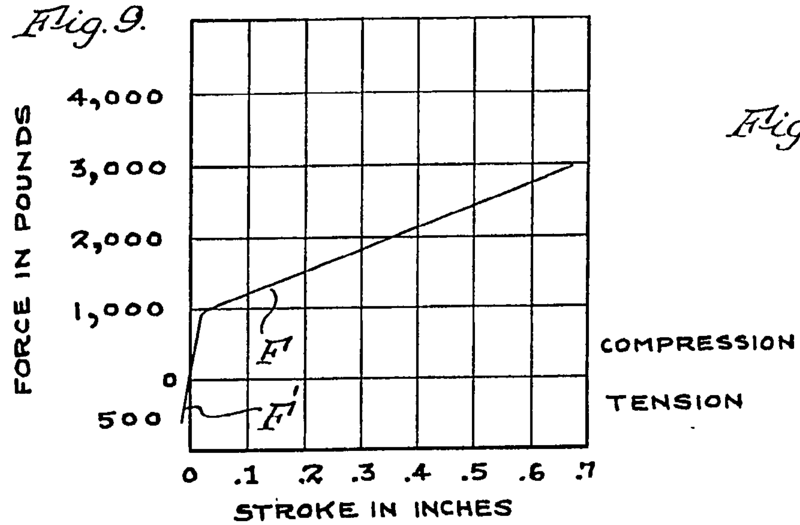


Fig. 13.

Fig. 9.



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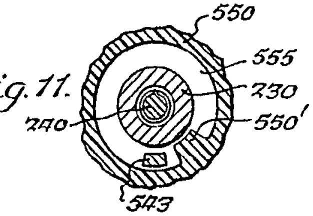

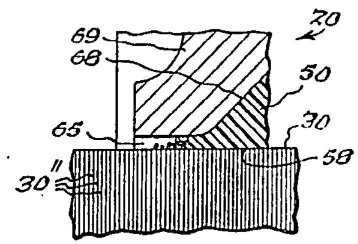

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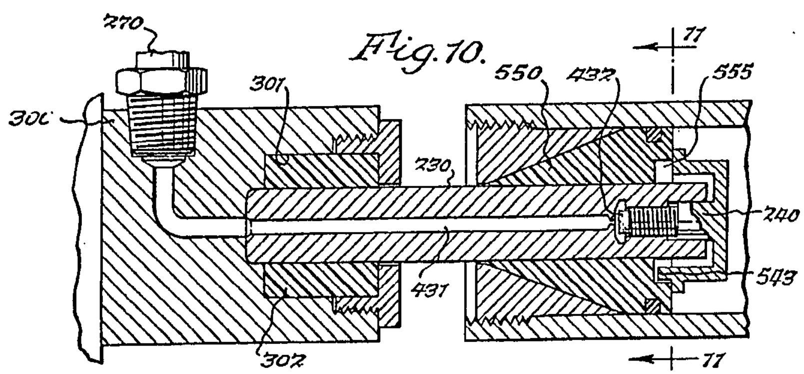

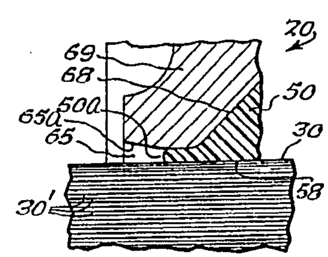

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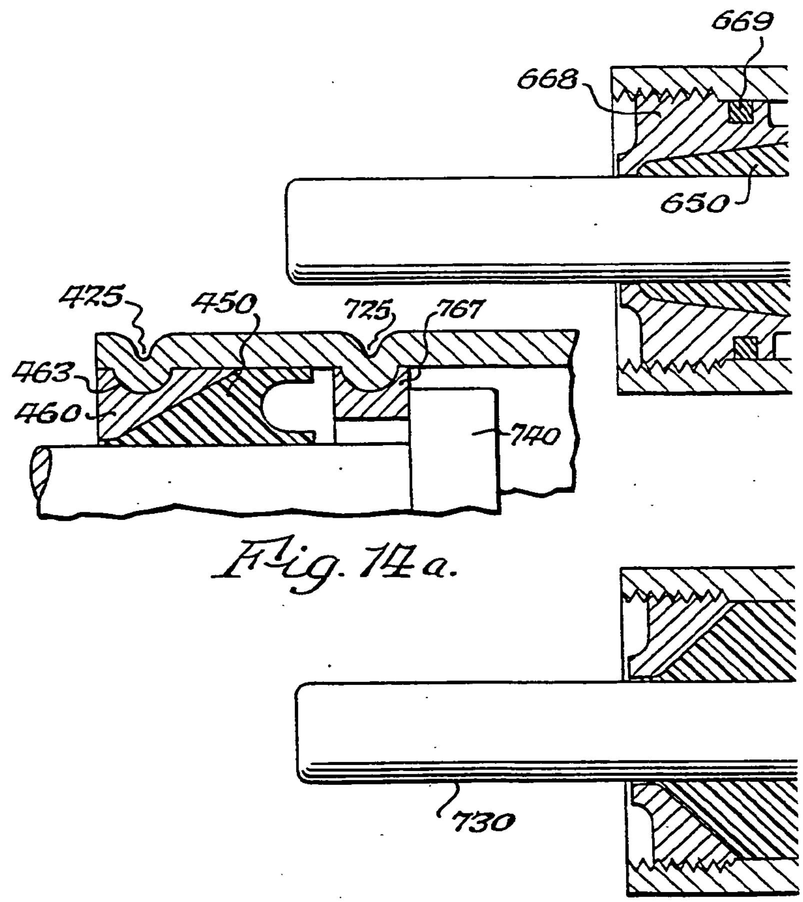

Fig. 14a.

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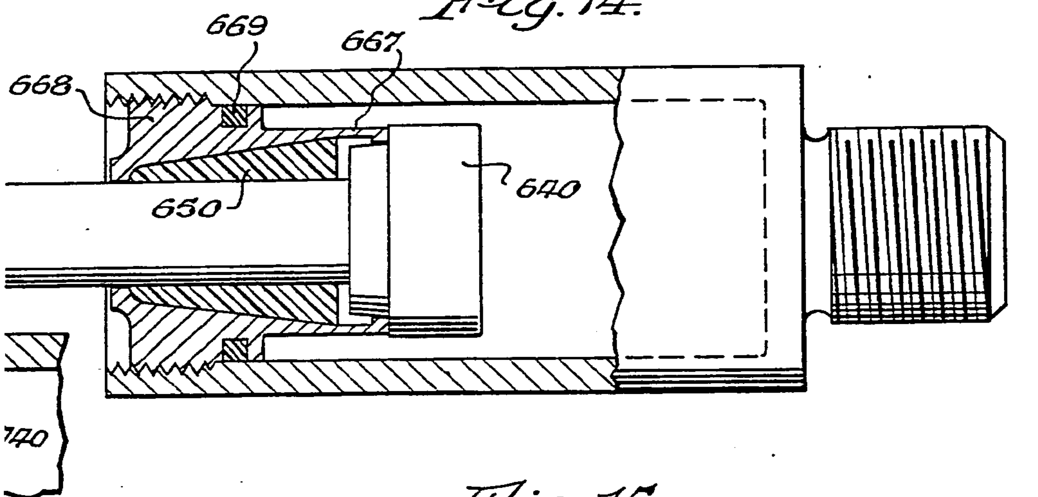
Sheet 5

Fig. 14.



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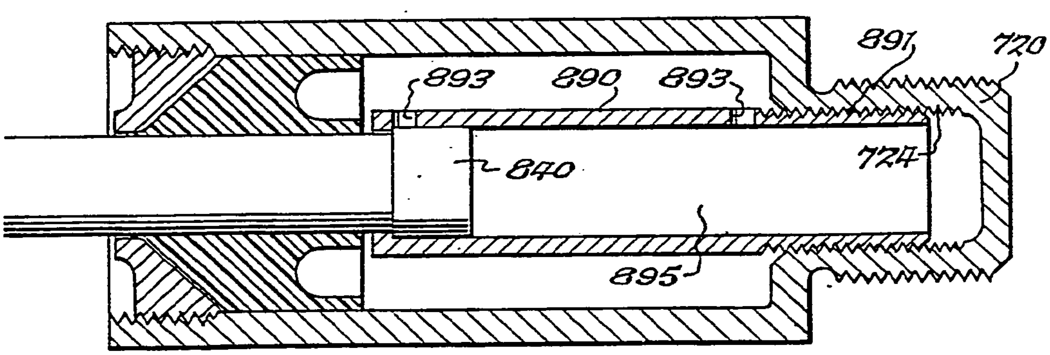

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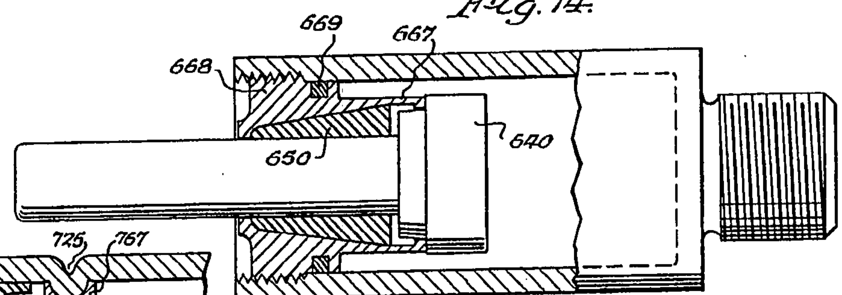

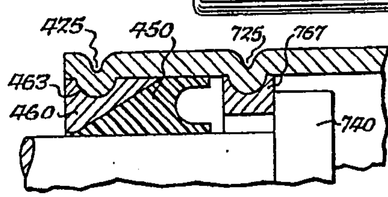

Fig. 14a.

Fig. 15.